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EFFICIENCY AUGMENTATION OF GAS TURBINE CYCLES

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ABSTRACT

The efficiency of gas turbine cycles can be enhanced by application of several methods. In the present contribution, the four most promising options e.g. compressor cooling, recuperation, reheating and elevation of turbine inlet temperature are discussed in detail. The potential of efficiency augmentation is depicted for all described methods with respect to the required effort. In addition, it is shown that the combination of different cycle improvement methods can give a disproportional high benefit because of upcoming synergy effects. For the compressor cooling it is worked out that an unconventional cooling by water injection gives a superior result over a conventional cooling. Furthermore, as any cooling of the compression process is accompanied with lower compressor outlet temperatures a strong potential for recuperation is provided by combining both methods. Finally, the obtainable efficiency of a gas turbine is determined for combination of several enhancement methods.

INTRODUCTION

Gas turbines are one of the most common devices in the field of power generation. Available sizes range from the so called Micro Gas Turbines (MGT) producing some Kilowatt of power to Heavy Duty Gas Turbines producing some hundred Megawatt of power. Today's designs reflect high running life at low investment for the high engine-power class. In addition, gas turbines are flexible with the operating mode and produce low emissions due to continuous combustion (NOx<10ppm, CO<15ppm @ 15 Vol.-% residual oxygen at full load) [1]. Therefore, usually no elaborate exhaust gas treatment is required [2]. A broad spectrum of liquid and gaseous fuels can be used to operate gas turbines which mostly tolerate changing fuel compositions. The elementary configurations of gas turbines which usually have only rotating parts inside comprise a compact design and power density accompanied by a good balance and low noise emissions. This potentially contains long intervals of maintenance (regular maintenance after 6.000 – 8.000 operating hours (oh); overhauling after 40.000 oh; life time expectation 80.000 oh) which means low maintenance costs.

Applications for high engine-power class gas turbines can be found usually in the electricity generation. Two cases of operation prevail all other applications: stand alone gas turbines for peak load and gas turbines in combined heat and power plant applications. The same employments are given for the MGT's while the medium size machines very often serve as movers for other machines.

In the last decade the use of medium and small gas turbines in the field of cogeneration of heat and power increased strongly [3]. Peripheral feed of electric energy into the grid has become a routine matter forced by the usage of renewable energy resources like wind, sun, organic substances and waste gases (landfill gas, mine gas, sewer gas, process gas) for example. The potential for local electric power supply is much bigger than the currently used portion. As the power output of a simple cycle gas turbine is only about one third mechanical energy and two thirds thermal energy this latter energy part of exhaust gases can be used to increase the efficiency of gas turbine cycles. For all gas turbines exhaust heat recovery is feasible at a level of about 650°C. Some examples for external waste heat recovery are: process energy, process steam, heating, Organic Rankine Cycles, etc.) [4], [5].

The before mentioned facts outline that simple gas turbine cycles are not very efficient. Improvements are strongly recommended in order to provide more opportunities for application of gas turbines. One method for example is internal heat recovery. Some of the known advancements of simple gas turbine cycles are specified as given in Table 1 [1], [6], [7].

Table 1 Advancements of simple gas turbine cycle

Recuperation	Reheating
Inlet-chilling	Catalytic combustion
Steam-cooled GT	Steam injection (STIG)
Spray-inter-cooling (SPRINT)	Humid Air turbine (HAT)
Inter-cooling (ICAD)	Chemically Recuperated GT

The present paper deals only with few of these methods. Out of the number of opportunities the most promising four have been selected and evaluated with respect to the required effort and the achieved result. In addition, combinations of the four chosen methods are also investigated.

NOMENCLATURE

a	kJ/kg_{Inlet}	Specific work
FS	kg_i/kg_{Fuel}	Fuel specification
h	kJ/kg	Enthalpy
H	kJ/kg	Fuel heating value
$\dot{m}$	kg/s	Mass flow
Ma	–	Mach number
p	Pa	Pressure
$\dot{Q}$	kJ/s	Heat flux
T	K	Temperature

Special characters

β	kg_{Fuel}/kg_{Air}	Fuel feed ratio
ε	–	Efficiency of heat exchanger
η	–	Efficiency
φ	–	Relative humidity
π	–	Pressure ratio
ξ	$kg_i/kg_{Mixture}$	Mass ratio of fluid composition mixture
ζ	–	Loss coefficient according to Levebre

Subscripts

0	–	Reference state
1	–	Inlet of gas turbine component
2	–	Outlet of gas turbine component
c	–	Cold side
$cold$	–	Cold pressure drop losses
C	–	Compressor
CC	–	Combustion chamber
DF	–	Diffusor
Fu	–	Fuel
h	–	Hot side
Hex	–	Heat exchanger
i	–	Inferior, index of gas turbine component index of fluid composition mixture
IC	–	Inter cooling
ID	–	Inlet duct
m	–	Middle
M	$kJ/kg \cdot K$	Molar mass
$\max$	–	Maximum
$mech$	–	Mechanical losses
$\min$	–	Minimum
Mix	–	Mixture
R	$kJ/kg \cdot K$	Universal gas constant
$real$	–	Real
s	$kJ/kg \cdot K$, –	Isentropic, superior
t	–	Total
T	–	Turbine
th	–	Thermal
$theor$	–	Theoretic
W	–	Water
WI	–	Water injection

GAS TURBINE CYCLE ASSESSMENT

Simple cycle as basis for comparison

Simple gas turbine cycles are characterised by the scheme depicted in Figure 1. Due to its simplicity and the limits concerning the technical boundary conditions the efficiency of the cycle is lower than 30% for small machines and not higher than 39% for big machines. Nowadays we are not able to afford such low efficiencies in the field of power generation.

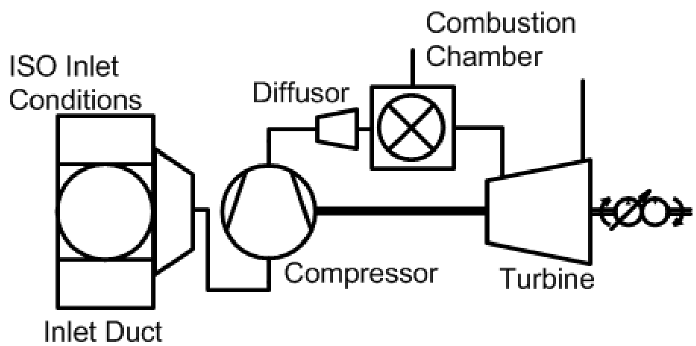

Figure 1 Simple gas turbine cycle

Independent of the method to improve the efficiency of such a basic cycle the included parts (inlet filter, compressor, combustion chamber and turbine) must be high efficient components. The thermal efficiency and the specific work output of the cycle can be calculated using the following equations:

$$\eta_{Cs} = \frac{h_{t2s} - h_{t1}}{h_{t2} - h_{t1}} = \frac{\Delta h_{Cs}}{\Delta h_C} \tag{1}$$

$$\eta_{CC} = \frac{\dot{m}_{Fu\ theor} \cdot H_s}{\dot{m}_{Fu\ real} \cdot H_s} = \frac{\dot{m}_{Fu\ theor}}{\dot{m}_{Fu\ real}} \tag{2}$$

$$\eta_{Ts} = \frac{h_{t2} - h_{t1}}{h_{t2s} - h_{t1}} = \frac{\Delta h_T}{\Delta h_{Ts}} \tag{3}$$

$$\eta_{th} = \frac{\sum_i a_i \cdot \eta_{mech}}{\beta \cdot H_i} \tag{4}$$

$$a = \sum_i a_i \cdot \eta_{mech} \tag{5}$$

Boundary conditions and fluid properties

On the basis of these well-known relationships describing the thermodynamic behaviour of gas turbine cycles a Mathcad based synthesis program has been created in dependence to Bräunling [8] to determine the thermal efficiency and the specific work output of the considered cycle. The simple gas turbine cycle is calculated using the boundary conditions depicted in figure 1.

The used fluid properties for the gas phase of dry and humid air as well as the combustion gases are provided by the formulation VDI 4670 [9] without dissipation at high temperatures. Whenever the pure substance water is in liquid state, the fluid properties calculated by the formulation IAPWS-IF97 [10] are considered. Using ideal gas properties as a basis for simple gas turbine cycle calculations is a common and improved method even at high combustion chamber inlet pressure due to concurrent increase and decrease of temperature in the compressor and turbine. For gas turbine cycles with a high amount of evaporated water inside the working fluid and lower temperatures at equal pressure levels this basis seems to be not appropriate because of the enlarged imbalance between ideal gas and real gas properties [11]. However for the calculations in this paper ideal gas properties are considered for the fluid in the gas phase and real fluid properties are considered for the water in liquid state. Further research based on real fluid properties for air [12] and water in the gas phase is in the focus of investigations to expose the influence on wet gas turbine cycles. An appropriate property library for humid air as real gas is allocated by Kretzschmar et. al. [13].

The thermal efficiency of the cycle as a function of compressor pressure ratio is shown in Figure 2 with the turbine inlet temperature as a parameter. It is very clear that the maximum efficiency increases strongly with increasing turbine inlet temperature. In addition, the maximum of the efficiency curve can be found at higher compressor ratios for higher turbine inlet temperatures. From Figure 2 it can be clearly seen that a high thermal efficiency of a simple gas turbine cycle can only be achieved with very high turbine inlet temperatures and high compressor pressure ratios at the same time.

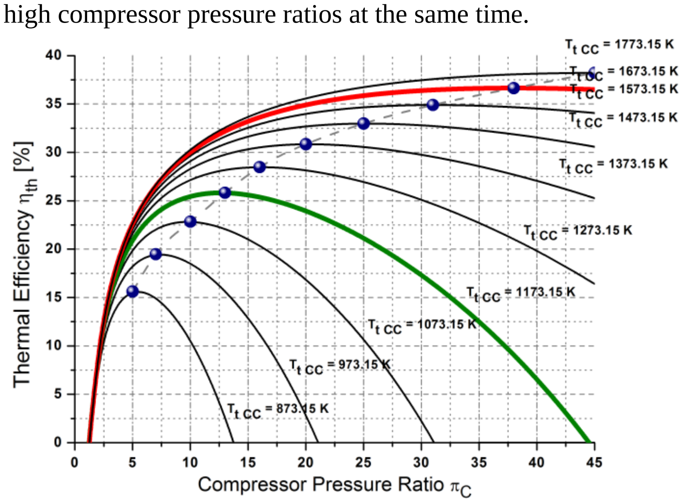

Figure 2 Thermal efficiency of simple gas turbine cycle

In Figure 3 the maximum attainable thermal efficiency and the turbine specific work output are plotted versus the compressor ratio for the turbine inlet temperature of $T_{t\,CC}$=1173 K which is assumed to be the maximum turbine inlet temperature without turbine cooling. The curves result from a variation of the isentropic efficiencies of compressor and turbine. The figure shows the curves for very high isentropic efficiencies of the compressor and of the turbine (η_{Cs}=88% and η_{Ts}=90%). For such high efficient machine components at a reliable turbine inlet temperature a maximum efficiency of 37.5 % can be achieved at a compressor pressure ratio π_C>20. It becomes also clear that the specific work is reduced from ≈ 230 kJ/kg at the operating point of highest specific work output at a compressor pressure ratio of π_C ≈ 9 to approximately 200 kJ/kg in the operating point of max. efficiency. This means the high efficient machine, which needs twice the compressor and turbine stages than the machine with maximum specific work, has a specific power output reduced by approx 20% compared to the machine designed for max specific work.

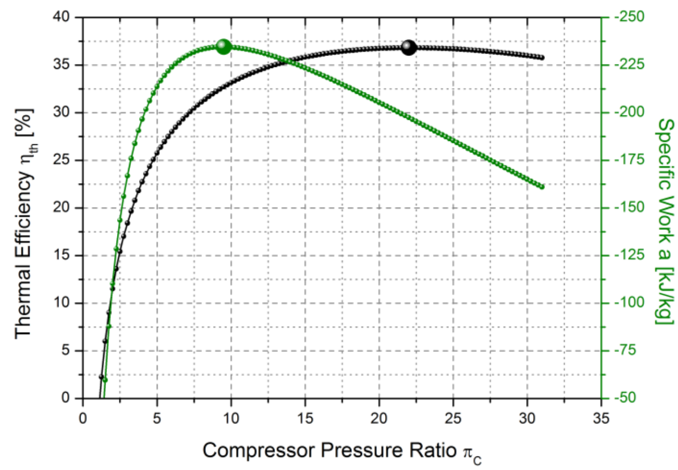

Figure 3 Attainable performance of simple GT-cycle

In the next steps four selected enhancement methods are described and evaluated in comparison to the simple cycle to enhance this limited efficiency. The focus of the investigation is directed to two different turbine inlet temperatures: the first one is $T_{t\,CC\,2}$=1173 K for which no turbine cooling is required; the second one is $T_{t\,CC\,2}$=1573 K which can only be realised with turbine cooling.

METHODS OF GAS TURBINE CYCLE ENHANCEMENT

Recuperation

Recuperation is an internal heat recovery method which transfers part of the exhaust energy to the compressed air before injecting fuel in the combustion chamber. The amount of transferred energy is a measure for the cycle improvement and it depends strongly on the heat exchanger efficiency. The heat exchanger efficiency can be determined using the following equations:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} \tag{6}$$

$$\dot{Q}_{max} = \min\left(\dot{Q}_h, \dot{Q}_c\right) \quad (7)$$

$$\dot{Q}_h = \dot{m}_h \cdot (h_{h1}(T_{h1}, \xi_h) - h_{h2}(T_{c1}, \xi_h)) \quad (8)$$

$$\dot{Q}_c = \dot{m}_c \cdot (h_{c2}(T_{c2}, \xi_c) - h_{c1}(T_{c1}, \xi_c)) \quad (9)$$

In an optimisation process the main parameters influencing the cycle efficiency (counter-current heat exchanger efficiency, heat exchanger pressure losses, compressor efficiency and turbine efficiency) have been varied [14], [15]. Both, increasing heat exchanger efficiency and decreasing heat exchanger pressure losses lead to rising cycle efficiency at strong reduced compressor pressure ratio. In this process first the maximum thermal efficiency is reached and then the maximum specific work is achived during an increase of the compressor pressure ratio. For high efficient compressor, heat exchanger and turbine (ε_{Hex}=90%; η_{Cs}=88% and η_{Ts}=90%) the thermal efficiency and the work output of the cycle are shown in Figure 4. The maximum efficiency of the recuperated cycle appears at a compressor pressure ratio less than π_C = 4. The specific work for this parameter combination is not very high but the aim of this investigation is to increase the efficiency and not to maximize the work output.

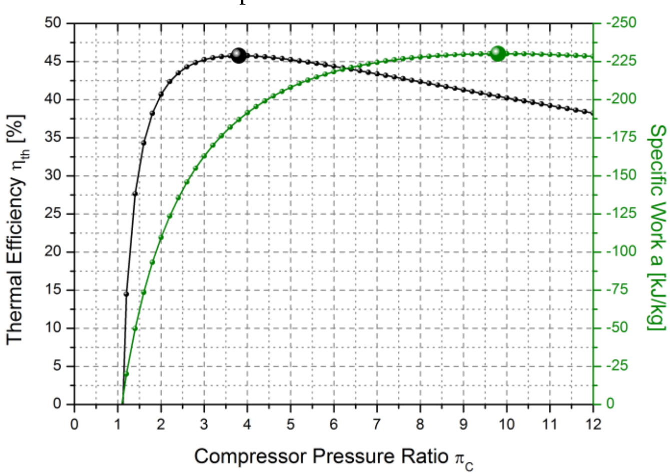

Figure 4 Attainable performance of GT-cycle with recuperation

Cooling of compression process (ICAD)

A recuperation process becomes much more reasonable when the temperature gap between the exhaust gas and the gas at the compressor outlet increases. As the turbine outlet temperature is only influenced by the efficiency of the expansion process in the turbine blades (for a give turbine inlet temperature) it is important to lower the compressor outlet temperature by cooling methods. Beside the advancement of recuperation cooling of the compression process gives the advantage of less required driving power for the compressor because of approaching an isothermal compression process. For example, cooling can be arranged in a conventional way with ordinary coolers after each compressor stage. In such a case a huge investment is required for the coolers and one has to deal with specific pressure losses in each cooler. In that way, energy is withdrawn from the process and conveyed to the ambiance usually with the help of another fluid. The process of heat exchange can be determined following the energy balance pictured in Figure 5.

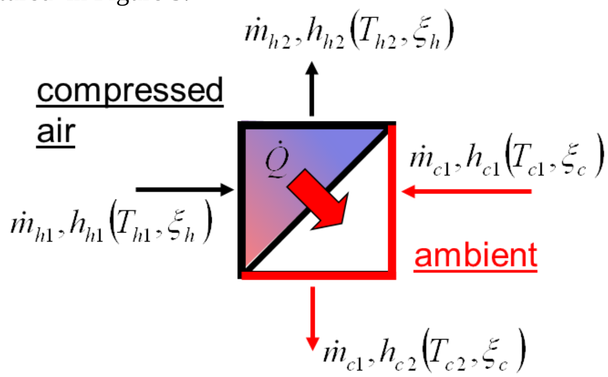

Figure 5 Energy balance of heat exchanger

Adopting a different cooling method which does not convey energy out of the process would be probably superior to the conventional cooling. Injecting water into the compressed air is a method which keeps all energy within the process and decreases the temperature of the compressed air by evaporation of the injected water at the same time. The process of water injection into the compressed air is depicted in Figure 6. To calculate this cooling method besides the air massflow and the heat flux coming from the compressor exit the additional massflow and heat of the injected water has to be taken into account. In this incection component of the process the evaporation of the water takes place. The phase change and mixture of the water steam with the air gives the thermodynamic state of the working fluid at the exit of this process component.

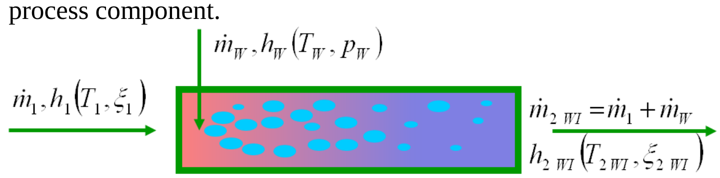

Figure 6 Calculation method for water injection

In Figure 7 a comparison of the two cooling methods is shown by sketching both cycles in a temperature-entropy diagram. Starting at the same thermodynamic state ($C1_{IC/WI}$) after polytropic compression of the air the state at the compressor outlet ($C2_{IC/WI}$) is also the same for both processes. At this point the two methods drift apart. The conventional cooling by an ordinary cooler takes energy out of the process and goes along the line of nearly constant pressure to a smaller entropy level. The entropy decreases mainly because of the change in temperature which is in dependence on the heat flux at constant fluid composition mixture described by the following equations (10) and (11).

$$\Delta s_{1/2}(T, p) = \sum_{i=1}^{8} \xi_i \cdot s_i(T_2, p_{i2}) - \sum_{i=1}^{8} \xi_i \cdot s_i(T_1, p_{i1}) + \Delta s_{Mix}(\xi_i) \quad (10)$$

$$\Delta s_{Mix}\left(\xi_i\right)=-\sum_{i=1}^{8} R_i\cdot\xi_i\cdot\ln\left(\frac{M_{Mix}}{M_i}\cdot\xi_i\right)=0 \qquad (11)$$

In contrast, a cooling by injection of water into the flow and complete evaporation at nearly constant pressure brings energy into the system and determining the mixing entropy gives higher entropy values defined by equation (12).

$$\Delta s_{Mix}\left(\xi_i\right)=-\sum_{i=1}^{10} R_i\cdot\xi_i\cdot\ln\left(\frac{M_{Mix}}{M_i}\cdot\xi_i\right)>0 \qquad (12)$$

This benefit proceeds for the other compressor stages and let the different cycles drift away from each other. The introduced energy due to the water injection is conveyed through the complete machine and preferably has to be utilised in a subsequent process (e. g. Organic Rankine cycle).

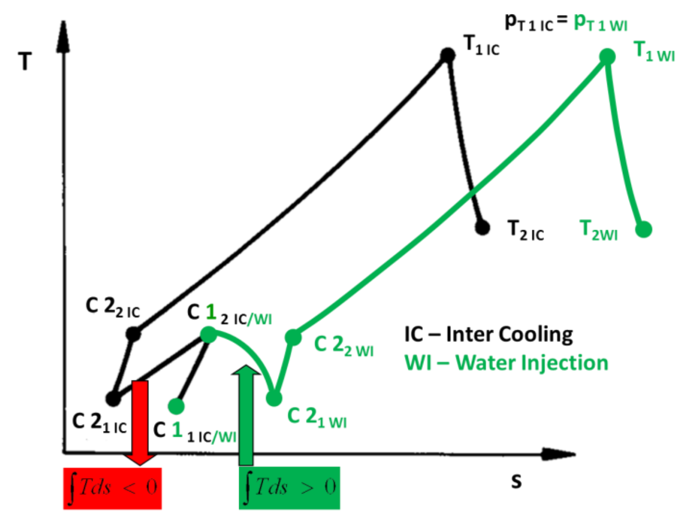

Figure 7 Comparison of conventional inter-cooling and water injection

Reheating

Reheat combustion has been proven to be a robust and highly flexible gas turbine concept for power generation. In the mid 1990s, Alstom introduced two similar sequential combustion gas turbines: the GT24 for the 60 Hz market and the GT26 for the 50 Hz market. The main technology differentiator of Alstom's GT24/GT26 gas turbines is the sequential combustion principle, which was already introduced in 1948 as a way of increasing efficiency at low turbine inlet temperature levels. The compressed air is heated in a first combustion chamber by adding about 50% of the total fuel (at base load). After this, the combustion gas expands through the high-pressure turbine, which lowers the pressure by approximately a factor of 2. The remaining fuel is added together with some additional cooling air in the second combustion chamber, where the combustion gas is heated a second time to the maximum turbine inlet temperature and finally expanded in the four stage low-pressure turbine [16].

The above described process of reheating using a second combustion chamber is shown in Figure 8 in comparison to a process featuring the single combustion. It can be taken from the qualitative temperature-entropy diagram that with two-stage combustion not only the converted energy is bigger than with a single combustion but also the averaged temperature of the added heat is higher. Looking at equation (10) this fact shows directly that the thermal efficiency is higher.

$$\eta_{th}=1-{T_0}/{T_m} \qquad (10)$$

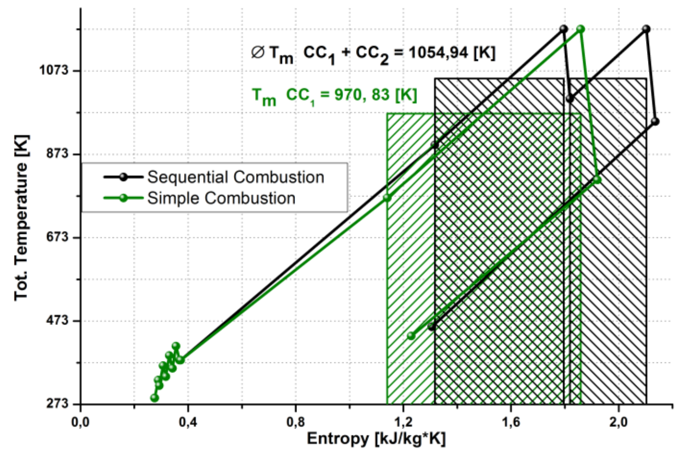

Figure 8 Benefit of sequential combustion

Elevation of turbine inlet temperature

Elevation of turbine inlet temperature is one method to increase the cycle efficiency. As shown in figure 2 not only the efficiency increases but the optimum of the efficiency is shifted to higher pressure ratios. This means that beside the high temperature a very high pressure inside the machine has to be handled. High turbine inlet temperatures up to more than 1773 K are state of the art in heavy duty as well as in aircraft turbines in combination with turbine cooling. Figure 9 gives an overview of the available turbine (air) cooling methods.

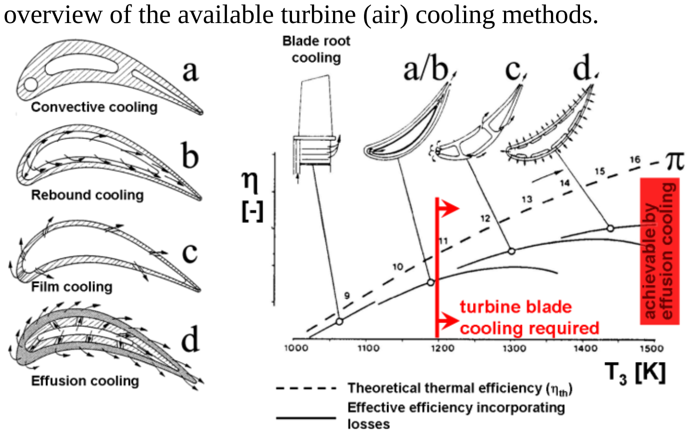

Figure 9 Comparison of cooling methods for expansion process

Utilising turbine inlet temperatures above 1173 K requires a cooling of the parts which are contacted by hot gas. High-grade cooling methods allow for higher turbine inlet temperatures. To go beyond the current limits new cooling methods are essential. Such a new method for example is the effusion cooling. The invention process for this cooling method is in progress. With this cooling method currently the highest $T_{t\,T\,1}$ can be reached because the flux of cooling air can be adapted exactly to the requirements [17], [18].

Turbine blade air cooling postulates an amount of air which is not guided to the combustion chamber and therefore does not participate at the energy input provided by the fuel. This fraction of air absorbs the required compressor work but is not energised by the fuel and therefore cannot perform the same work as the energised fluid does. In this way the gain of efficiency due to increased turbine inlet temperature has to be regarded in contrast to the energy deficit due to the use of not energised fluid. Within a certain limit of turbine inlet temperature, the benefit for the cycle is higher than the effort for turbine cooling. For this range of turbine inlet temperatures an increasing efficiency of the cycle can be observed and elevation of turbine inlet temperature is a reasonable method to increase the cycle efficiency.

TOTALLING OF CONSIDERED EFFECTS

The examination of the above specified methods shows that the gas turbine cycle efficiency can be enhanced by applying every single method. The occurring synergy effects when combining several enhancement methods will be discussed in the following chapter. As mentioned before, two different turbine inlet temperatures are investigated. The lower inlet temperature can be applied without turbine cooling while the higher T_{in} requires severe cooling. The boundary conditions for the performed cycle calculations are depicted in the following Figure 10 to Figure 13 and Table 2.

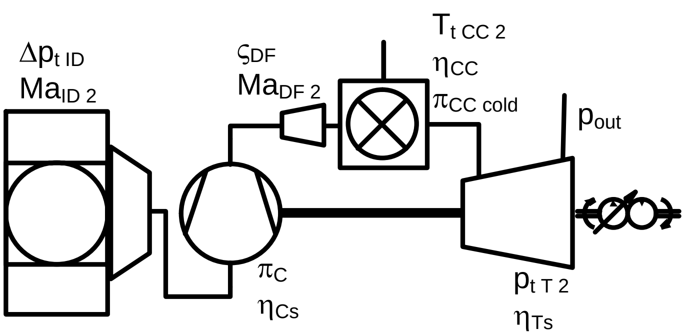

Figure 10 Simple cycle conditions

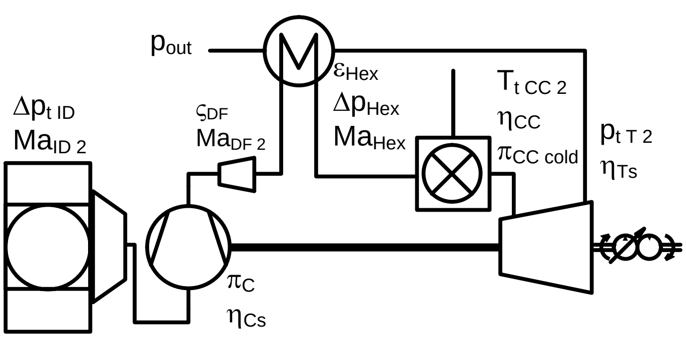

Figure 11 Recuperative cycle conditions

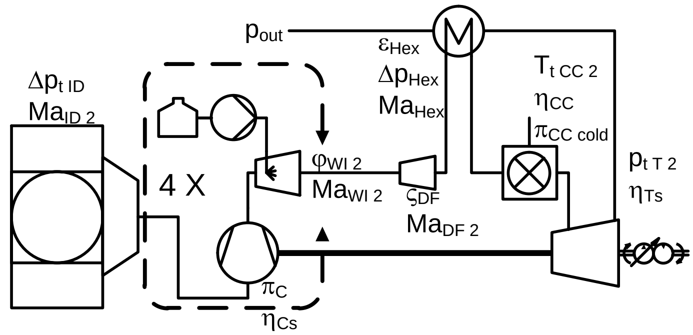

Figure 12 Recuperative + WI cycle conditions

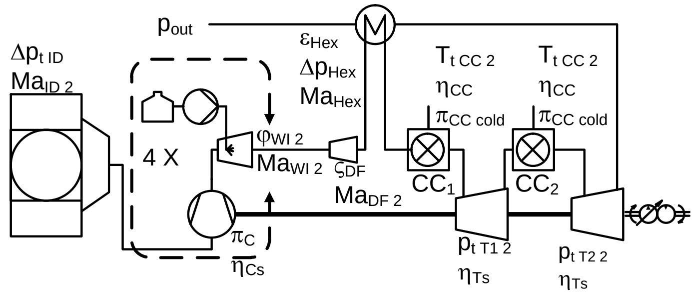

Figure 13 Recuperative + WI + 2 X CC cycle conditions

Table 2 Boundary conditions of simulated cycles

Part	Variable/Unit	Simple cycle	Recuperated cycle	Recuperated cycle and water injection	Recuperated cycle, water injection and sequential combustion
Definition of ambient conditions	p_t [Mpa]	0.1013	0.1013	0.1013	0.1013
	T_t [°C]	15	15	15	15
	φ [%]	60	60	60	60
Adiabatic inlet nozzle	Δp_{t_NB} [kPa]	0.222	0.222	0.222	0.222
	$Ma_{NB\ 2}$	0.4	0.4	0.4	0.4
Compressor	π_C	22	3.8	5.9	9
	η_{Cs}	88	88	88	88
Isobaric water injection	$p_{t\ W}$ [MPa]	-	-	15	15
	$T_{t\ W}$ [°C]	-	-	20	20
	η_{pump} [%]	-	-	60	60
	φ_2 [%]	-	-	50	50
	$Ma_{WI\ 2}$	-	-	0.4	0.4
Adiabatic diffusor	ζ_{DF}	0.15	0.15	0.15	0.15
	$Ma_{DF\ 2}$	0.1	0.1	0.1	0.1
Counter-current heat exchanger	ε_{Hex} [%]	-	90	90	90
	Δp_{Hex} [kPa]	-	1	1	1
	$Ma_{Hex\ 1}$	-	0.1	0.1	0.1

Combustion chamber	FS	Natural gas Thyssengas GmbH (October 2009) [19]			
	η_{CC} [%]	99.8	99.8	99.8	99.8
	$\pi_{CC\ cold}$	0.98	0.98	0.98	0.98
	$T_{t\ CC\ 2}$ [°C]	900	900	900	900
Turbine	$p_{t\ T\ 2}$ [MPa]	0.1013	0.1023	0.1023	0.4887
	$p_{t\ T2\ 2}$ [MPa]	-	-	-	0.1023
	η_{Ts} [%]	90	90	90	90
Mechanical efficiency	η_{mech} [%]	100	100	100	100

The basis of all comparison is the simple cycle which has been depicted already earlier. As mentioned before, applying of a recuperated process leads to much higher cycle efficiency at a tremendously smaller compressor pressure ratio as shown in Figure 14. For the assumed boundary conditions the compressor pressure ratio of the recuperated process is below $\pi_c = 4$. As another benefit, the transferable specific energy is almost the same and it remains at nearly the same pressure ratio. The efficiency enhancement and the reduction of the compressor pressure ratio are the most important facts regarding the recuperated process. Exploiting these effects, most of the small gas turbines available at the market utilise the exhaust energy for recuperation of the compressed air.

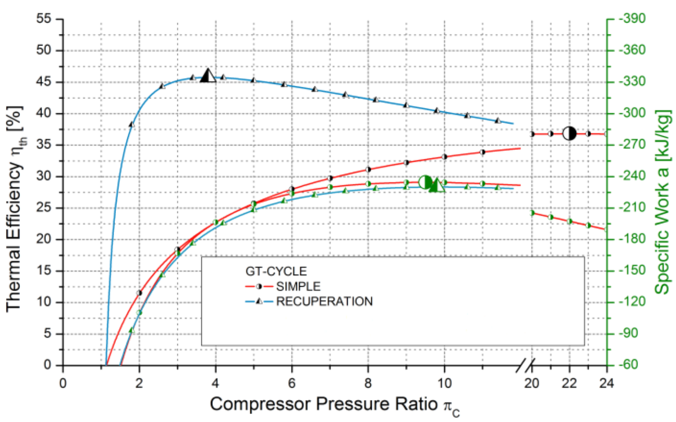

Figure 14 Comparison of simple cycle and recuperated cycle

Application of cooling of the compression process using water injection after every compressor stage in combination with recuperation leads to a further increased attainable efficiency at an only little higher compressor ratio. As can be taken from Figure 15 the maximum of the cycle efficiency is higher than 40% at a compressor ratio of about $\pi_C = 6.5$. Another positive effect of cooling by water injection is the strongly increased specific work of the machine. For the same pressure ratio the transferable specific work is about 50% higher than without water injection.

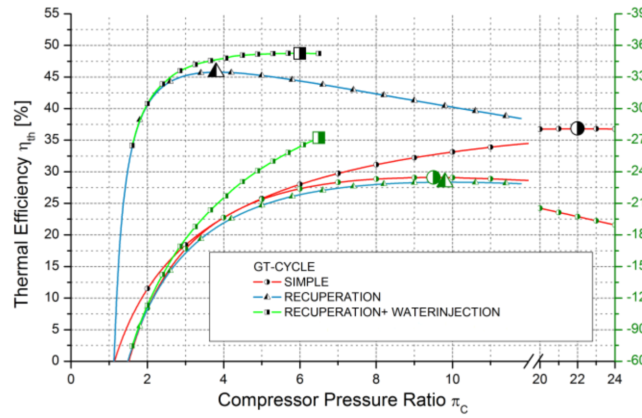

Figure 15 Comparison of simple cycle and recuperated cycle with cooling by water injection

The combination of recuperation and cooling by water injection creates a strong efficiency augmentation of the gas turbine cycle in comparison to the simple cycle. A further increase of efficiency is hard to achieve. Accounting for reheating gives another potential of efficiency enhancement. But reheating makes sense only if the turbine is a multi-stage machine. A sequential combustion is known only for heavy duty gas turbines in the class of hundreds of Megawatts. In this paper, the fragmentation of the fuel energy into two equal parts is investigated. As shown in Figure 16, reheating gives a minor increase of cycle efficiency when combined with recuperation and cooling by water injection. The efficiency of the gas turbine cycle can be increased over 50% at a compressor pressure ratio of about $\pi = 9$. But another fact which should not be neglected is that the transferable energy increases strongly. At the maximum of the cycle efficiency the transferable specific energy is the highest of all compared processes. With this severe density of the energy conversion process small and compact machines can be fabricated.

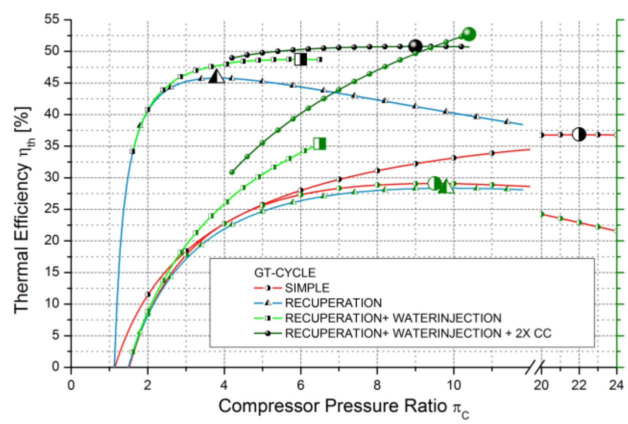

Figure 16 Comparison of simple cycle and recuperated cycle with cooling by water injection and reheating by sequential combustion (T_{T1} = 1173 K)

The last examined method of gas turbine efficiency enhancement is the elevation of turbine inlet temperature. As already shown in Figure 2 a strong efficiency enhancement can be observed with increasing turbine inlet temperature. But in

Figure 9 the technical limits of this parameter are depicted. As the utilised materials cannot withstand a temperature higher than 1173 K on a continuing basis turbine cooling methods must be applied in conjunction with increased turbine inlet temperature. For turbine cooling in most cases air from the compression process is used. The amount of cooling air is dependent on the gas turbine operating point and therefore variable. For the investigation performed for this paper a constant amount of cooling air is assumed and the turbine inlet temperature is fixed to T_{T1} = 1573 K.

Again the different methods of cycle efficiency augmentation are applied to the cycle. In principle the same procedure than already explained for the lower turbine inlet temperature takes place. Every single method is able to increase the efficiency of the cycle and combination of the regarded methods gives a superior behaviour of the gas turbine process.

In Figure 17 the combination of all four efficiency enhancement methods are applied to a gas turbine cycle with a turbine cooling. Again, recuperation and compressor cooling by water injection leads to a strong efficiency increase whereas water injection and reheating serve for much higher energy density by increasing the transferable specific work. Concluding, a cycle efficiency of about 60% can be achieved together with a specific work which is twice the specific work of a cycle with low turbine inlet temperature (Figure 16).

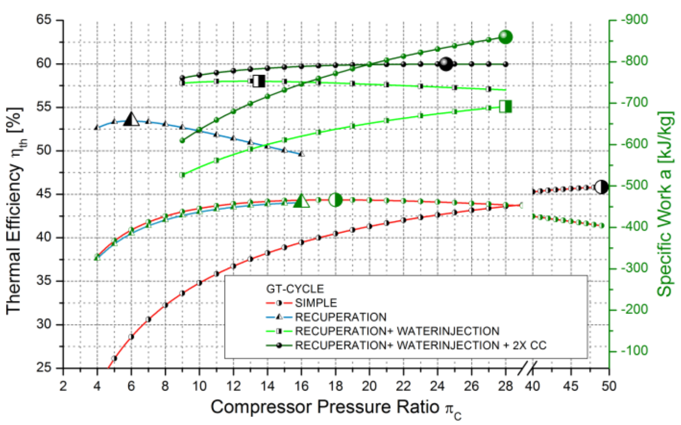

Figure 17 Comparison of simple cycle and recuperated cycle with cooling by water injection and reheating by sequential combustion (T_{T1} = 1573K)

CONCLUSIONS

Numerical investigations of gas turbine cycles have been performed in order to quantify different methods of efficiency augmentation. Four promising enhancement methods are discussed in detail and applied to the gas turbine process. Overall, the results show that every single method has its potential to increase the cycle efficiency. The combination of several methods exhibits an additional effect because of reclaimable synergy effects. Especially the combination of compressor cooling and recuperation is reasonable because of a much bigger temperature gap between turbine exhaust gas and compressed air. Elevation of turbine inlet temperature in general stands for higher thermal efficiency of a gas turbine process and in combination with other improvement methods the impact on the efficiency is more distinctive. In contrast to the efficiency enhancement methods other procedures preferably increase the transferable power density and therefore are dedicated to make machines small at a given power transfer rate. As shown from the numerical calculations water injection into the compression process seems to be a method which raises not only efficiency but also specific power output of a gas turbine. Reheating (sequential combustion) gives only a slight increase in efficiency but reduces the emissions and raises the specific power output enormously.

From the performed study it can be learned that gas turbines can have a thermal efficiency higher than 50% even with a turbine inlet temperature of 1173 K. The technical effort due to combination of recuperation, compressor cooling by water injection and sequential combustion is much higher than for a simple cycle gas turbine but the gain in efficiency is almost more than 15 percentage points.

Additional application of turbine blade cooling allows for higher turbine inlet temperatures and consequently leads to a further increased thermal efficiency of 60% when again combining the regarded enhancement methods.

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